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Study of Strength Requirements for Nozzles for Ice Transiting Ships

Summary Report

Prepared for
Transportation
Development Centre

Transport Canada

JULY 1985

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TP 6838E
STUDY OF STRENGTH REQUIREMENTS FOR NOZZLES FOR ICE TRANSITING SHIPS
SUMMARY REPORT

PREPARED FOR
TRANSPORTATION DEVELOPMENT CENTRE

BY: V. LASKOW
C. REVILL

CANADIAN MARINE DRILLING LTD.
JULY 1985

The contents of this report reflect the views of the contractor and not necessarily the official views or policies of Transport Canada.

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17. Abstract

The objective of the project was to define and study in detail the factors and design criteria affecting ice-going propeller nozzle strength requirements and to formulate general analytical expressions which, when calibrated, could be used for design and regulatory purposes.

The major effort in the study of nozzle strength was the development of an analytical model. Bottom, side and axial loading cases were investigated for a ring structure. Bending and torsional moments, shear and tangential forces and deflections were calculated around the ring. In addition the foundation reactions and the effect of load spread were determined. The applicability of these results to a three-dimensional case was confirmed by finite element modelling.

The study concluded by suggesting requirements for regulatory body consideration and by proposing a full-scale test program.

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17. Résumé

L'objectif de ce programme était de définir et d'approfondir les facteurs et les critères de calcul concernant la résistance à donner aux tuyères protégeant les hélices des navires naviguant dans les glaces, et d'etablir des équations d'application générale visant à faciliter les fonctions de calcul et de réglementation.

L'objectif principal de la présente recherche a été la formulation d'un modèle tenant compte des régimes de charge s'exerçant dans les trois axes d'une structure annulaire. Les moments de flexion et de torsion, les efforts de cisaillement et les forces tangentielles, ainsi que les déformations correspondantes ont été calculés. En outre, les réactions dans le carlingage ainsi que l'effet de la répartition des charges ont été déterminés. Enfin, l'applicabilité du modèle formulé à une structure tridimensionnelle a été confirmée par la méthode des éléments finis.

Le rapport termine par des suggestions concernant les critères de réglementation et en préconisant un programme d'essais en vraie grandeur.

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1. PROJECT OBJECTIVE

The project objective was:

"To define and study in detail the factors and design criteria affecting ice going propeller nozzle strength requirements and to formulate general analytical expressions which, when calibrated, can be used for design and regulatory purposes."

2. OVERVIEW

Prior to investigating the structural strength aspects of the nozzles, a number of areas were studied to determine the range of nozzle parameters to be investigated (in particular diameter and power). Upon completion of these studies the structural strength was investigated using both analytical and finite element models. Findings from the investigations were incorporated in regulatory recommendations.

The following sections summarize this work.

3. REVIEW OF THE STATE-OF-THE-ART

A bibliography search identified only a few papers dealing with the strength of nozzles under hydrodynamic loading. Only Lloyd's Register of Shipping offers guidelines for nozzles of low powered ships; yet the number of icebreaking ships outfitted with shrouded propellers is steadily growing. In the Canadian Arctic 12 ships have shrouded propellers. Details of the ships to which the project had access are shown in Table 1. The largest nozzles are fitted to the M.V. Arctic (5.4 m diameter and 14,400 bhp) and the M.V. Canmar Kigoriak (5.33 m diameter and 16,400 bhp). As calculated in this study the nozzle of the Kigoriak can withstand loads of up to 1,400 tonnes radially and 3,200 tonnes axially, before reaching yield.

SHIP	VESSEL DISPLACEMENT TONNES	HORSE-POWER/ SHAFT	NOZZLE DIAMETER, m	PLATING THICKNESS, cm	NR OF RING WEBS	SECTION MODULUS, cm^3	CALCULATED MAXIMUM FORCE, TONS	ASPPR ICE CLASS
Kigoriak	7000	16400	5.33	3.5	3	49,320	1427	3
Robert LeMeur	5000	4500	3.03	2.5	3	13,125	1000	3
Supplier 1 & 2	2300	3500	2.8	1.27	3	8,757	387	2
Supplier 7	1774	2500	2.6	1.25	4	6,613	189	2
Arctic	36000	14400	5.4	2.5	2	24,713	350	2

CANADIAN MARINE DRILLING LIMITED

STRENGTH REQUIREMENTS FOR NOZZLES FOR ICE TRANSITING SHIPS

TITLE: NOZZLE PROPERTIES OF EXISTING SHIPS

TABLE 1

4. RANGE OF SHIPS AND POWERS INVOLVED

2 m, 4 m, 6 m, and 8 m diameter nozzles were selected to cover the whole range of present and future applications. The principal structural characteristics of each of the nozzles are shown in table 2. The 8 m diameter nozzle was chosen for each shaft on a twin shaft icebreaking tanker with a total horsepower of 100,000 and a displacement of 300,000 tonnes. Based on current design trends the relationship between horsepower and the disk area of shrouded propellers was established as:

Disk Area $\approx$ $(HP)^{0.88}/232$, where disk area is in m^2, or

Disk Area $\approx$ $(KW)^{0.88}/177$ if power is in kilowatts.

NOZZLE	VESSEL DISPLACEMENT TONNES	NO. OF PROPS	SHP PER PROP	PROP DIA. METERS	NOZZLE MATER'L	PROFILE LENGTH CM	PROFILE THK'S CM	SHELL THK'S CM	NUMBER OF WEBS	WEB THK CM	LONG'L SPG CM	AREA X-X CM^2	AREA Y-Y CM^2	TOTAL AREA CM^2	MODULUS X-X CM^3	MODULUS Z-Z CM^3
1A	2000	2	2000	2	EH36	100	20	2.5	4	2.5	30	200	500	700	5666	15000
1B	2000	2	2000	2	EH36	108	25	2.5	4	2.5	30	250	540	790	7791	18720
1C	2000	2	2000	2	EH36	116	30	2.5	4	2.5	30	300	580	880	10200	22813
2A	10000	2	8000	4	EH36	200	40	2.5	4	2.5	30	400	1000	1400	22666	60000
2B	10000	2	8000	4	EH36	216	50	2.5	4	2.5	30	500	1080	1580	31166	74880
2C	10000	2	8000	4	EH36	232	60	2.5	4	2.5	30	600	1160	1760	40800	91253
3A	30000	2	20000	6	EH36	300	60	3.6	4	3.6	30	864	2160	3024	73440	194400
3B	30000	2	20000	6	EH36	324	75	3.6	4	3.6	30	1080	2333	3413	100980	242611
3C	30000	2	20000	6	EH36	348	90	3.6	4	3.6	30	1296	2506	3802	132192	295660
4A	300000	2	50000	8	EH36	400	80	4.2	5	4.2	30	1680	3360	5040	156800	393000
4B	300000	2	50000	8	EH36	432	100	4.2	5	4.2	30	2100	3629	5729	216440	488073
4C	300000	2	50000	8	EH36	464	120	4.2	5	4.2	30	2520	3898	6418	284256	593734

Notes:

1. Area X-X denotes effective shear area of vertical members of nozzle profile section.
2. Area Y-Y denotes effective shear area of horizontal members of nozzle profile section.
3. Number of webs includes leading and trailing edge plating.
4. Modulus X-X denotes section modulus about longitudinal axis of the nozzle section.
5. Modulus Z-Z denotes section modulus about vertical axis of the nozzle section.
6. All numbers are based on the simplified profile having rectangular shape.

CANADIAN MARINE DRILLING LIMITED	STRENGTH REQUIREMENTS FOR NOZZLES FOR ICE TRANSITING SHIPS	TITLE: PRINCIPAL STRUCTURAL CHARACTERISTICS OF THE NOZZLE SERIES TABLE 2

5. BASIC NOZZLE GEOMETRY, LOCATIONS AND ATTACHMENTS

Nozzles are rings made of airfoil type sections. Most of the nozzles have the following characteristics:

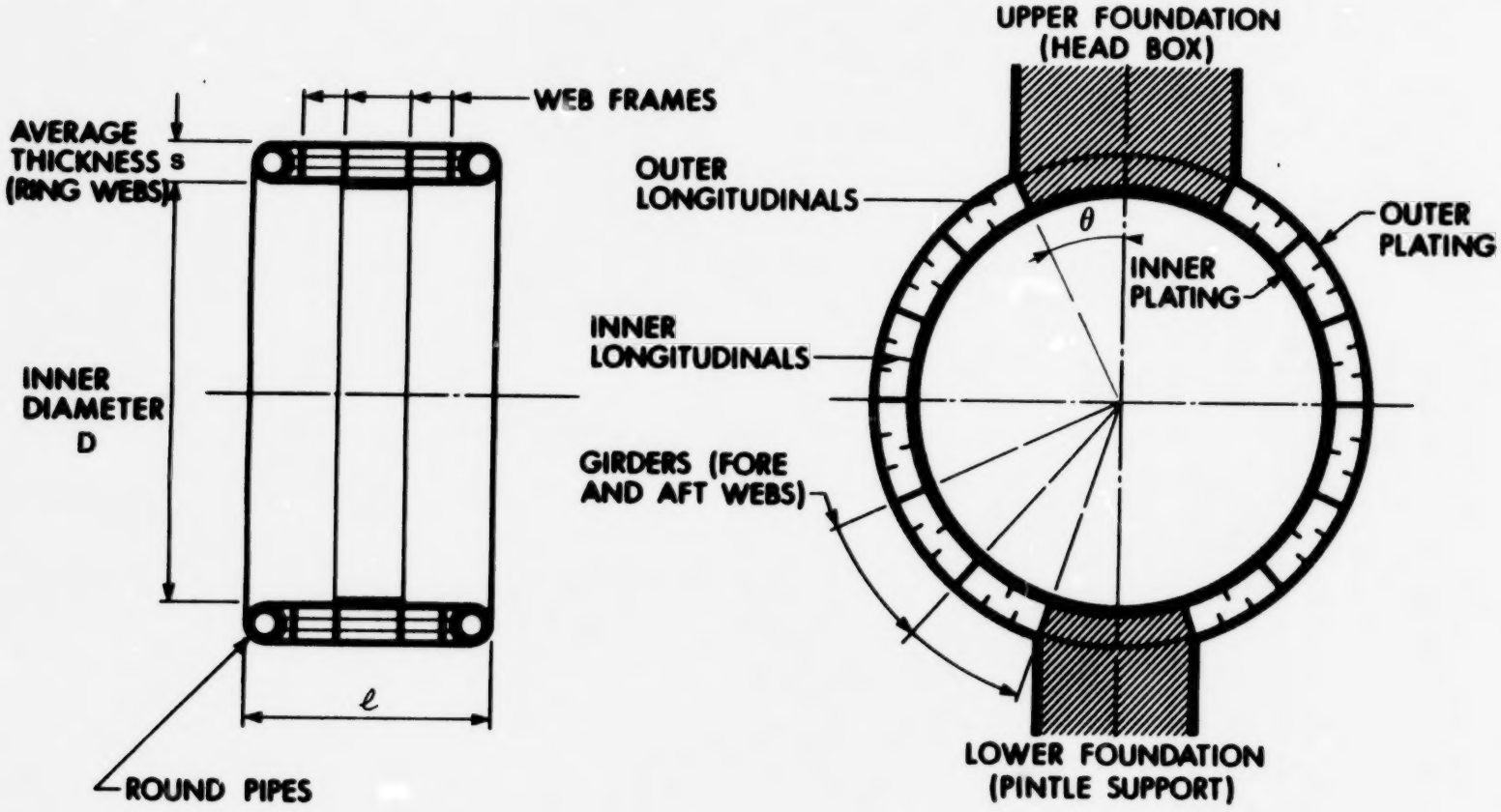
Nozzle Length/Diameter	0.5	<	l/D	<	0.8
Profile Thickness/Length	0.07	<	s/l	<	0.19
Entrance Angle	10.2°	<	α_i	<	15.2°
Camber	0.03	<	f/l	<	0.07

The most commonly used nozzles are NSMB #19A and #37, which have simplified shapes.

Structurally nozzles are composed of:

- inner and outer plating, with the supporting stiffeners,
- rings webs,
- girders,
- upper/lower foundation,
- leading/trailing edge structure (pipe/castings).

Figure 1 illustrates these features.

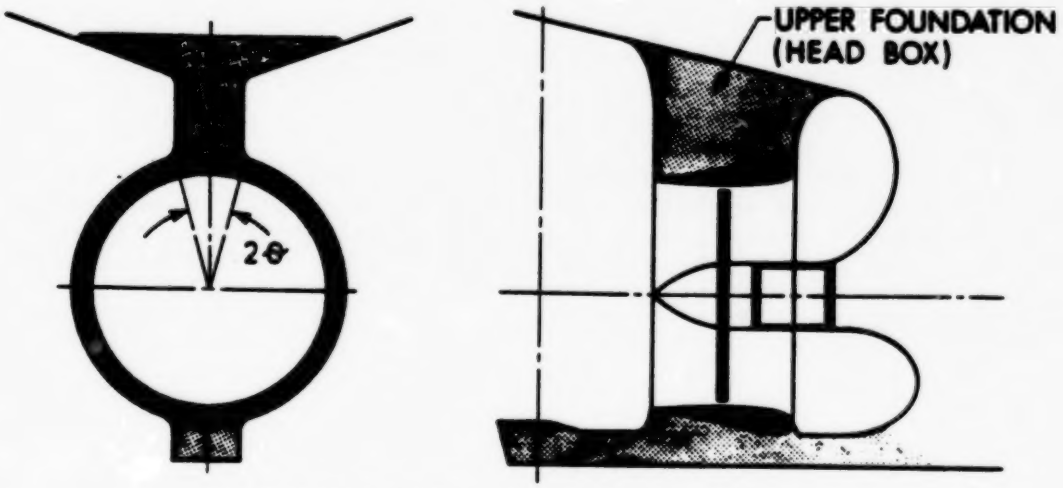


CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transitting Ships Summary Report	TITLE: Schematic Structural Configuration Of The Nozzle
		FIGURE No. 1

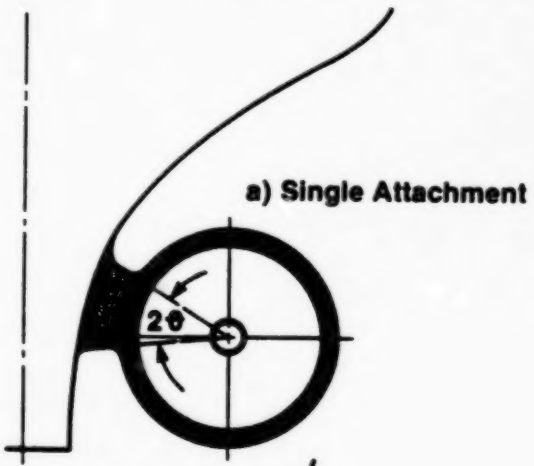
The upper limit for the section moduli (Z) of the existing nozzles can be expressed as:

$$Z\ (cm^3) \approx 1000 \times D^{2.35}$$

The upper foundation angle Θ (defined as one half the angle over which the nozzle is attached at its upper surface to the ship's structure) varies between 8.5° (Arctic) and 90° (Robert LeMeur). The various methods of nozzle attachments to the hull are shown in figure 2.

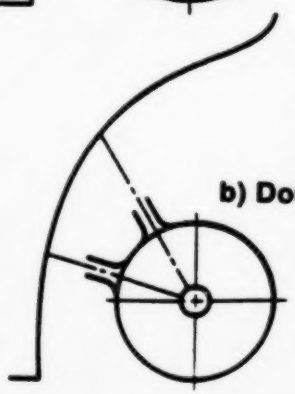


OPEN CENTRAL LOCATION



SEMI-TUNNEL
CENTRAL LOCATION

OPEN WING



CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transitting Ships Summary Report	TITLE: Nozzle Attachement To The Hull
		FIGURE No. 2

6. RANGE OF ICE FEATURES

Bergy bits, growlers, multiyear ridges and first year ice were features identified as potentially interacting with nozzles. The first three were considered to have low mobility and to have an "upper bound" crushing strength of 5 MPa. These features can be positioned on the side or the bottom of the nozzle.

7. INTERACTION SCENARIOS

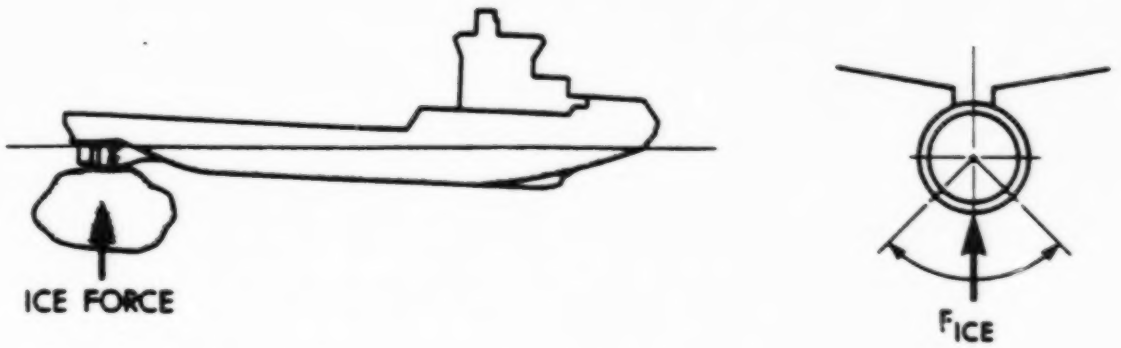
Three scenarios have been identified below and are shown schematically in figure 3:

- grounding on ice;
- swaying into ice; and
- ramming ice.

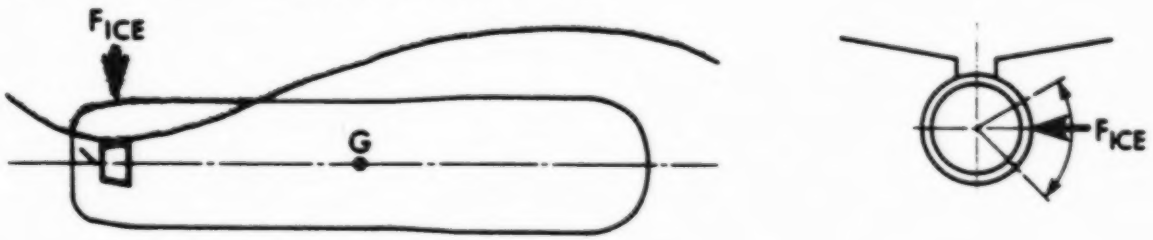
Simple dimensional analysis has been shown to be inadequate to predict the range of ice impact forces. A computerized ship-ice interaction simulator (ICESIM) was run for a limited number of ice/ship interaction scenarios on two ships (30,000 tonnes and 300,000 tonnes displacement) with different nozzle sizes and differing nozzle locations on the smaller vessel. The main conclusions were:

- The magnitude of the maximum ice force grows with increased ship displacement;
- The magnitude of the maximum ice force grows with increased ice crushing strength;
- Due to the central location of a single shaft nozzle, the vessel's rolling response to a load is not as great as the response of a ship with twin shaft nozzles in a similar nozzle/ice interaction. Single shaft nozzles can therefore generate higher force than twin shaft nozzles when installed on vessels of similar displacement. However due to the more protected location of a single centreline nozzle, the loads are expected to be lower than on a ship with twin nozzles;
- Side impacts generate higher forces than bottom impacts.

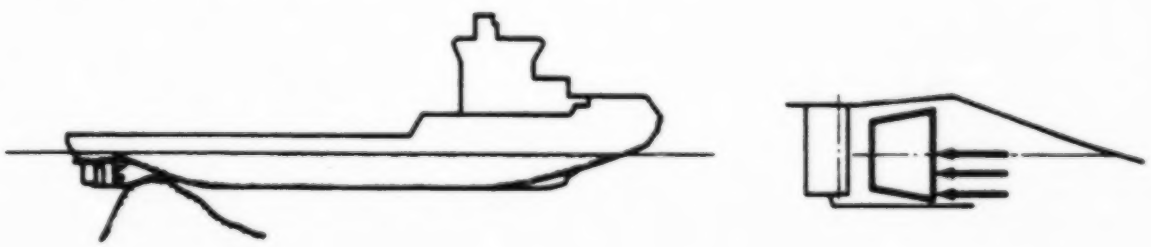
SCENARIO 1 - GROUNDING-ON-ICE



SCENARIO 2 - SWAYING-INTO-ICE



SCENARIO 3 - RAMMING-INTO-ICE



CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transiting Ships Summary Report	TITLE: Basic Interaction Scenarios
		FIGURE No. 3

8. ANALYTICAL MODEL

Analytical modelling was selected as being most suitable for the general investigation of nozzle strengths. A series of nozzles was designed covering the whole range of possible applications. The basic nozzle shape selected was NSMB #37. Three (3) different profile thicknesses/nozzle length ratios were established:

$s/\ell = .2$,
$s/\ell = .23$, and
$s/\ell = .26$

An investigation of the influence of s/ℓ on the hydrodynamic performance was performed. No significant deterioration of nozzle-propeller performance was found with increasing nozzle thickness.

The influence of shear and hoop force on the deformations of the nozzle ring were found to be small. Consequently, a thin-curved-beam theory was applied to solve for:

- bending moments;
- shear forces;
- hoop forces, and
- deflections;

as a function of nozzle geometry, and applied load.

Two "base" cases were investigated in detail:

- side impact, and
- bottom impact.

For unit loads, an infinitely stiff top foundation, no bottom foundation, and equal inner and outer plating thicknesses, the principal findings were:

- Side impacts produce more severe stresses than bottom impacts.

- Maximum stresses in the nozzle occur at that side of the foundation which is nearest to the load application point.

- Bending moment stresses are larger than those due to shear and hoop stresses.

- The highest stresses occur in the inner ring plating. These stresses are compressive.

- When the nozzle inner plating is stressed to yield value, the nozzle ring deflections lie within acceptable limits (1% of nozzle radius).

- Increase of foundation angle reduces the bending stresses significantly.

- Spreading of the load around the perimeter does not significantly reduce the maximum bending stresses, when compared to the stresses from an equivalent concentrated load.

- The highest stresses were caused by a load application at 105° measured from bottom centre. These stresses did not vary significantly for foundation angles from 0° to 20°.

- A unit axial load was found to produce much lower stresses than a unit radial load.

- Two expressions have been developed to estimate the force (F_{max}) in tonnes which causes yielding of the inner ring plating at the foundation. If the nozzle structure has been designed and the section modules is known, then the following expression can be used:

$$F_{max} = Z \times \sigma_y / (R \times 1.04\ (.645 - (\theta°/213))$$

Where:

Z = the section modulus of nozzle,
R = the mean radius of the nozzle,
$\theta°$ = the foundation 1/2 angle ($\theta \leq 90°$),
σ_y = the yield strength of the material.

All units must be consistent.

At the preliminary design stage, only general nozzle particulars may be known and the following expression can be used.

$$F_{max} = D \times t_{pl} \times \sigma_y \times (471/(137 - \theta°)) \times ((\ell/D)/(1+D/s)$$

Where:
t_{pl} = the plating thickness of the ring (assumed constant),
ℓ = the length of the profile,

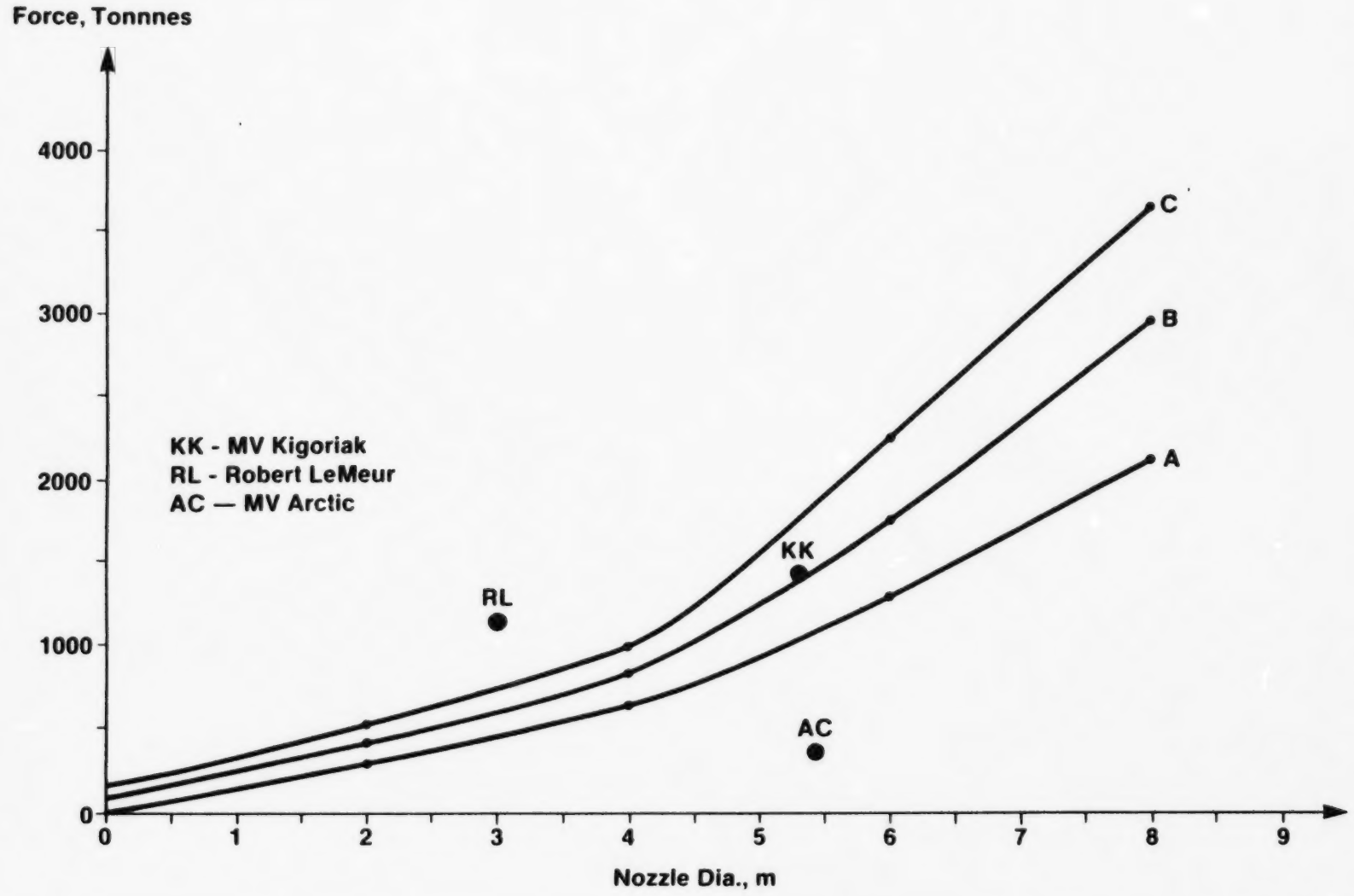
D = the diameter of the nozzle,
s = the profile thickness.

All units must be consistent.

Figure 4 shows the strength of the designed nozzle series as a function of the nozzle diameter.

- A brief assessment of nozzle's vibratory behaviour has shown no danger of resonance with blade-forced vibrations.

- A brief assessment of buckling strength of the nozzle has shown no danger of global buckling of nozzle structure.



CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transiting Ships Summary Report	TITLE: Comparison Of Radial Strength Of The Nozzle Series FIGURE No. 4

9. FEM MODEL

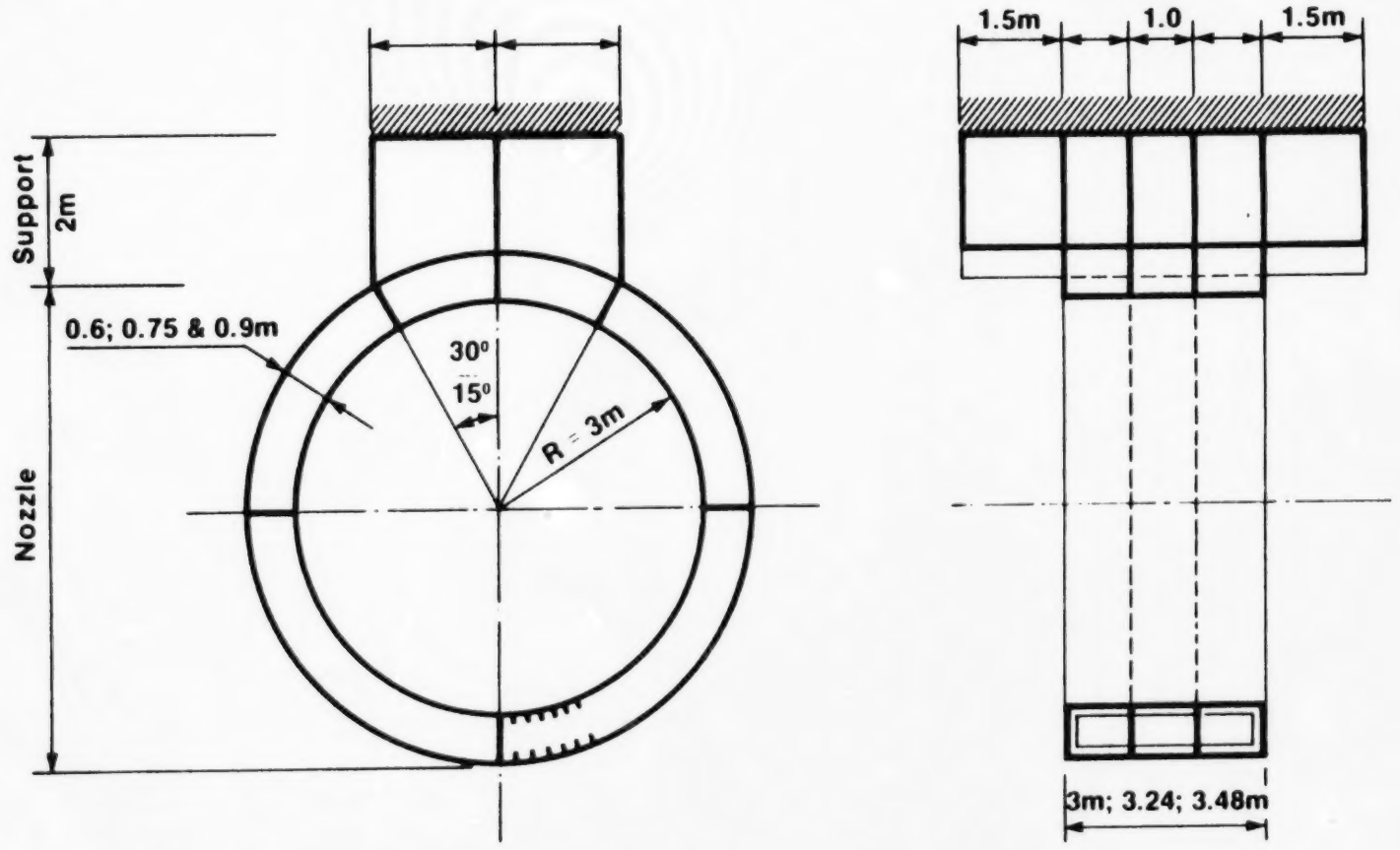
A FEM model of a 6 m diameter nozzle and its foundation was constructed. This model is shown in figure 5. The variation of key parameters in several runs lead to the following conclusions:

- The analytical model based on thin curved beam theory compares well with the FEM predictions for the bending moments, shear forces and tangential forces everywhere in the ring except at the connection to foundation.

- The reduction of bending moment, (compared to the analytical model) at that point is of the order of 10%. A comparison of the bending moment distribution between the finite element and analytical models is shown in figure 6.

- Neither the thickness of the profile nor the radius of the nozzle were found to influence the bending moment coefficients (expressed as the ratio of the bending moment at given location to the force times radius of the nozzle).

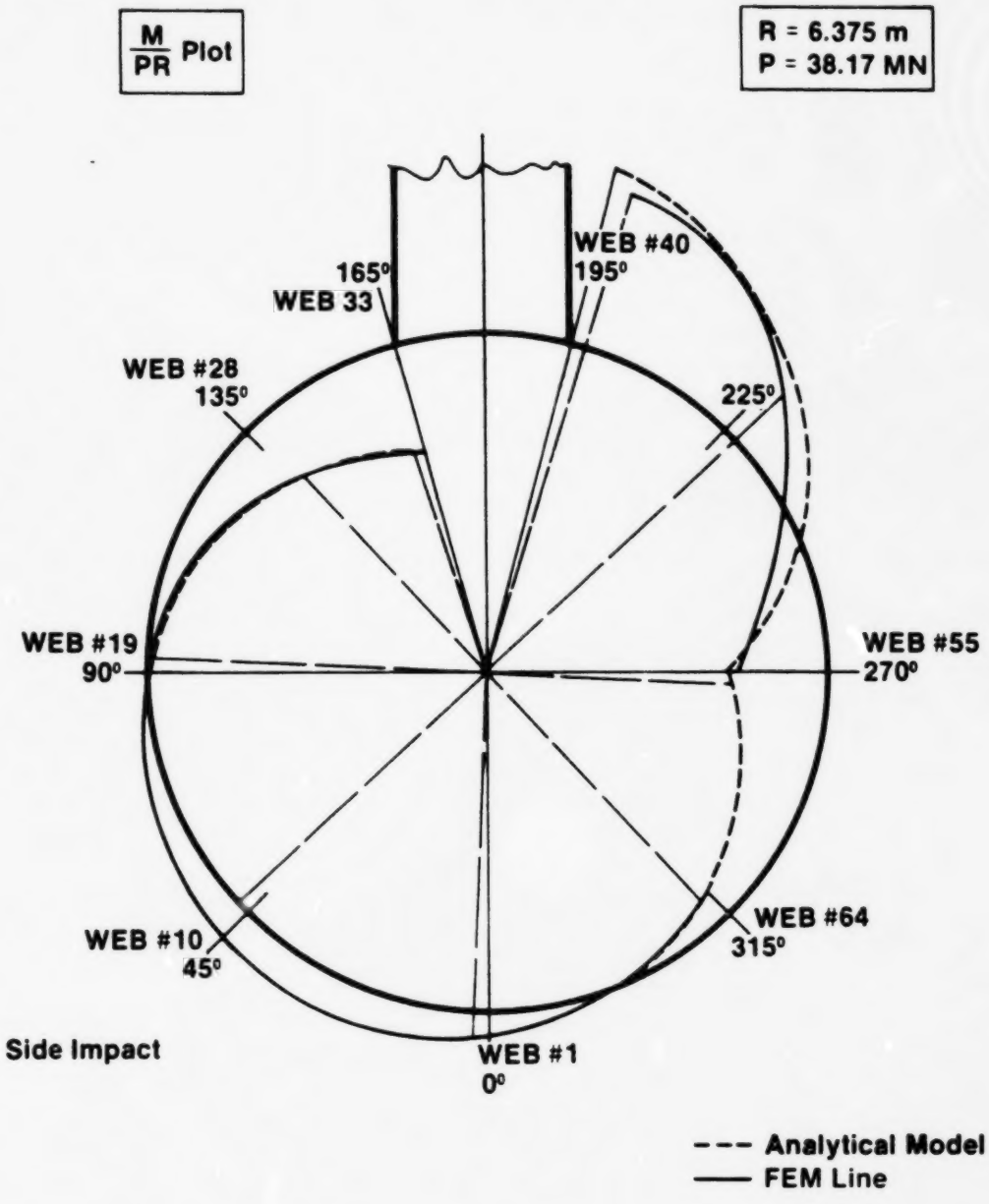
- Deflections of the nozzle ring were found to be significantly larger than those predicted by the analytical model, due to the influence of foundation deformation and local plating deformations. Foundation chosen for FEM modelling was not stiff enough to support the load.



Outer Skin { Nozzle: PL 36 mm, Long'ls 200 x 36 F.B., 300 SPG, 1000 Span

Support: PL 27 mm, Long'ls { 300 x 27 F.B., 300 SPG, I.W.O. 1500 Span
200 x 27 F.B., 300 SPG, I.W.O. 1000 Span

CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transiting Ships Summary Report	TITLE: Scantlings Of The Nozzle For FEM Investigation FIGURE No. 5



CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transitting Ships Summary Report	TITLE: Comparison Of Bending Moment Distribution Between FEM And Analytical Model
		FIGURE No.6

10. SUGGESTED REQUIREMENTS FOR REGULATORY BODY CONSIDERATION

a) Global Strength

Two parallel approaches are recommended:

Design to Nominal Force

When a designer has complete freedom in choosing the shape and scantlings of nozzle, the only requirement is that the nozzle must be able to withstand an ice force, being expressed as:

$(F)_{ice} = K_1 \times K_2 \times (\Delta HP)^{\beta}$ (see paragraph 10.3 of volume 1, final report)

Where:

K_1 = nozzle location factor,

K_2 = ice class factor,

Δ = the displacement of the ship, in tonnes x 10^{-3}

$(F)_{ice}$ = design ice force, in tonnes.

HP = rated horsepower x 10^{-3}

Tentative values, subject to the future full scale calibration are:

K_1 = 1 for single shaft ships

K_1 = 1.2 for twin shaft ships

If megawatts are used in above formula, then

K_1 = 1.04 for Single Shaft Ships

K_1 = 1.25 for Twin Shaft Ships

Tentative values of K_2 are calculated from the following equation:

K_2 = 100 + 60 x ICE CLASS (ASPPR)

Coefficient β is set at 0.137 and is also the subject of future calibration.

Examples of the value of F_{ice}, in tonnes, are given in the following table:

ICE CLASS 2

HP	Δ 2,000	4,000	10,000	20,000	30,000
5,000	301	332	376	413	437
10,000	332	365	413	454	480
15,000	350	385	437	481	508
20,000	365	401	455	500	528

ICE CLASS 10

HP	Δ 30,000	60,000	90,000	300,000
30,000	1,777	1,955	2,066	2,437
60,000	1,955	2,150	2,272	2,680
90,000	2,066	2,272	2,402	2,833

Design to Required Strength Parameters

1. Minimum global strength section modulus is:

$$Z \geq K \times F_{ice} \times R \times 1.04 \times (0.645 - \Theta/213°)/\sigma_y$$

Where:

Z is the section modulus, expressed in cm^3,
K is the safety factor, tentatively set at 1.0,
F_{ice} is the ice force, defined as above,
$\Theta°$ is the foundation 1/2 angle, in degrees,
σ_y is the yield strength of the material in tonnes per cm^2,
R is the effective nozzle radius, in cm measured from the shaft centreline to the midweight of the profile.

2. For steels having $\sigma_y \leq 360$ MPa no special considerations have to be given to the deflections of the nozzle ring. For higher yield strength steels, the minimum required inertia of the cross-section is:

$$I_{min} \geq \times 22.8 \times F_{ice} \times R^3/ER_o$$ (see paragraph 10.4 of the final report)

R_o is the radius of the inner plating in cm, E is the elastic modulus of the steel (GN/m^2) and I_{min} is expressed in cm^4.

The above formulation is used for $\Theta = 15°$. For other values of Θ, see Chapter 8 of Volume 1.

3. The minimum required shear area in the radial direction is:

$$A_{shear} \geq 0.9 \times F_{ice}/\tau$$

Where τ is the shear strength of the material.

4. The shear area of the nozzle attachment to the foundation in the axial direction should not be less than that calculated from the formula:

$$(A_{shear})_{axial} > F_{ice}/\tau$$

This formulation incorporates a safety factor of 2. Where possible, the area should be equally distributed between the upper and lower foundations.

b) Local Strength

Outer plating and stiffeners should be calculated using the ice pressures for the stern, multiplied by a safety factor:

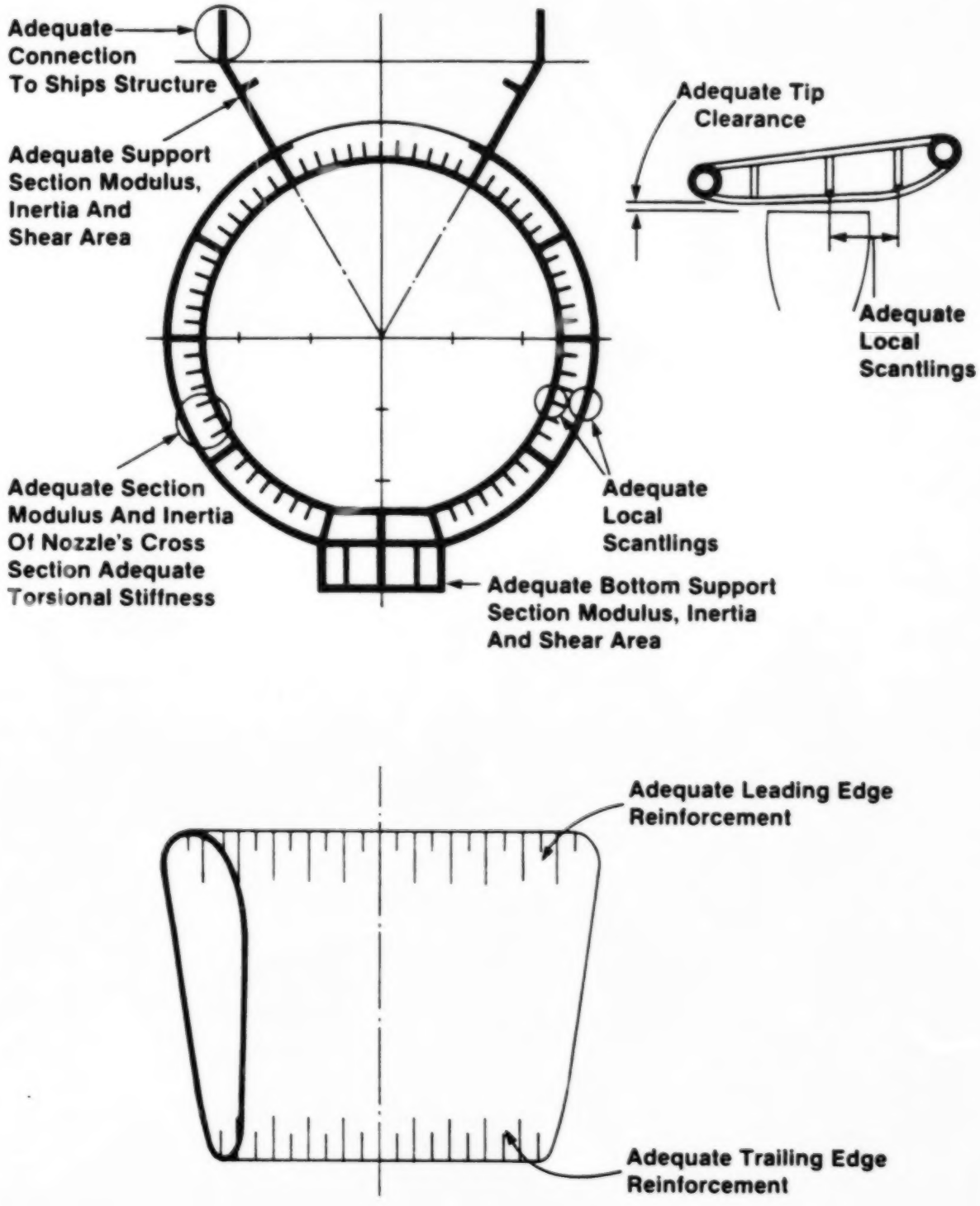
$$P_{nozzle} = 1.5 \times P_{stern}$$

Foundation Support

The nozzle should be attached at the top and bottom to the stern structure. Unless the bottom support is substantial, it should only be included in the assessment of nozzle strength when subject to an axial load.

The stiffness of the foundation should be such as to limit the foundation induced displacements of the ring. Subject to further investigation of the effects of foundation stiffness, these foundation induced displacements should be limited to 25% of the total ring displacement for side impacts.

Global and local strength requirements are shown schematically in figure 7.


Adequate Connection To Ships Structure
Adequate Support Section Modulus, Inertia And Shear Area
Adequate Tip Clearance
Adequate Local Scantlings
Adequate Section Modulus And Inertia Of Nozzle's Cross Section Adequate Torsional Stiffness
Adequate Local Scantlings
Adequate Bottom Support Section Modulus, Inertia And Shear Area
Adequate Leading Edge Reinforcement
Adequate Trailing Edge Reinforcement

CANADIAN MARINE DRILLING LIMITED	Strength Requirements For Nozzles For Ice Transiting Ships Summary Report	TITLE: Global And Local Strength Requirements
		FIGURE No. 7

11. PLANNING OF FULL SCALE TEST TO CALIBRATE THE ANALYTICAL MODEL

A comprehensive research program is proposed in order to quantify the following expression for one ship:

$$F_{max} = K_1 \times K_2 \ (\Delta \ HP)^{\beta}$$

It was felt that satisfactory instrumentation of an existing nozzle would be extremely difficult and expensive. Therefore only a limited instrumentation of an existing nozzle is recommended in order to gather information about nozzle stresses. A test program has been developed for a new ship with a high ice class (at least Class 4). The program would include the following steps:

- Design of the nozzle to the requirements of the proposed regulations.
- FEM modelling of the nozzle and foundation to understand the structural response to various loading locations and to determine the instrumentation location.
- Ship instrumentation and calibration.
- Full scale trials.
- Data analysis.

12. COMPUTER PROGRAMS

A series of programs were developed enabling any user to assess the strength of the designed nozzle.

The programs calculate the following:

Bending moments, shear forces, tangential forces, foundation reactions for any nozzle and direction of load application. A point or distributed load can be used.

Strength of any nozzle can be assessed by inputing actual structural properties of the nozzle.

13. CONCLUSIONS

1. The nozzle strength increases with the increase of its diameter. The rate of increase depends on the plating thickness and geometry characteristics.

2. The nozzle strength increases with the increase of profile thickness and/or lengthening of the profile. Hydrodynamic performance is not significantly affected by either.

3. The governing stresses are compressive stresses due to bending and tangential force. The inner plating will yield in compression, before the outer one.

4. Side loads cause higher stresses than the bottom loads or axial loads of the same magnitude.

5. The largest stresses occur near the foundation support.

6. The analytical model predicts the stresses well, except for the area near the foundations, where lower stresses occur due to the foundation flexibility.

7. Deformations of the nozzle ring are not significant factors in strength calculations for infinitely stiff foundations. For very weak foundations however, they are decisive.

8. Nozzles of 2 m diameter can withstand radial loads of approximately 400 tonnes. Nozzles of 8 m diameter can withstand radial loads of 3,000 tonnes, for the foundation angle $\Theta = 15°$.

9. ICESIM model has been proven to be usefull in predicting the general dependancies of ice load on the ship displacement, number of shafts and ice strength.. The absolute results cannot be used for predictive purposes without full scale calibration.

10. If full scale test results prove that loads are higher than those recommended in the proposed rules, the difference can be easily accommodated by the increase of the foundation angle, plating thickness and profile thickness and/or length. For example, using nozzle 4C with plating 0.050 m and $\Theta = 60°$, the strength of the nozzle becomes 5850 tonnes.

14. RECOMMENDATIONS

Based on the present study the recommended procedure for calculating any nozzle's strength requirements is given below.

The inputs, assumed to be available, are:

- Ship displacement and horsepower
- ASPPR ice class
- Nozzle diameter and section profile
- Upper foundation angle $\Theta°$,
- Nozzle location (centre/wing).

The steps for calculating the nozzle strength are as follows:

1. Calculate ice force, using the formula:

$$F_{ice} = K_1 \times K_2 \times (\Delta HP)^{\beta}$$

2. Calculate nozzle plating thickness using the formula for ice pressure:

$$P_{nozzle} = 1.5\ P_{stern}$$

3. Calculate nozzle profile section modulus using simplified (rectangular box) profile.

4. Calculate nozzle strength using the formula:

$$F_{max} = Z \times \sigma y/R \times 1.04\ (.645 - \Theta/213)$$

5. Compare F_{max} with F_{ice}. If $F_{max} < F_{ice}$, increase any of (or combination of):

   - Foundation angle
   - Plating thickness
   - Profile thickness
   - Strength of material

6. Check shear area in radial and longitudinal directions as per paragraph 10-4.

7. Check other requirements, including:

   - Bottom attachment
   - Moment of inertia
   - VR of webs

Detailed list of the requirements is given in Chapter 10 of Volume 1 of the main report.

8. If optimization of the nozzle structure is contemplated, use the programs developed for detailed evaluation of the stresses in the nozzle ring.

Recommended Future Studies

A study of the influence of the structural characteristics of nozzle top and bottom support on the nozzle response to ice loads is recommended.

A study to optimize the nozzle structure by varying the section properties around the circumference (especially at the foundation) to match the stress distribution should be carried out.

A study of the influence of the side velocity of impact and pressure area distribution on the magnitude of ice impact forces on the nozzles should be carried out. Although a comprehensive program as set out in section nine is necessary in the longer term, we recommend a reduced scope of measurements on the MV Kigoriak to gather statistical data on nozzle stresses over a season of operation. The results would be compared to the results from a finite element model of the nozzle and its supporting structure to determine the loading on a nozzle operating in a wide range of ice conditions.